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Advanced performance of an open desiccant cycle with internal evaporative cooling

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Abstract

An enhanced evaporatively cooled open sorption cycle which can be driven by solar thermal heat is presented as an environmentally benign alternative to conventional air conditioning cycles. Experimental data of the core component - the desiccant coated heat exchanger - cooled by water evaporation is given. This evaporation of water in the heat exchanger leads to an increase in adsorbed water mass by 46 % and an enhancement of the cooling capacity by factor 4.1 when compared to a solely air cooled process. The newly designed heat exchanger is then compared by a simulation analysis to data of an air cooled open sorption process published in earlier research. It is shown that the desiccant mass is used more efficiently in the more recent prototype. Further considering the evaporation in the heat exchanger underlines the more advanced performance that can be achieved with the new concept and prototype design.

Keywords: Desiccant; cooled sorption cycle; sorptive coated heat exchanger

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1. Process description and objectives

Due to an increasing desire for comfort coupled with rising temperatures, not at least due to heat islands in urban areas [1], the demand for domestic air conditioning is expected to rise significantly [2]. Applying conventional, electrically driven compression chillers to satisfy the cooling and dehumidification loads may therefore lead to higher electricity consumption and increases the risk for grid failures in summer peak hours. Especially in climates characterized by high latent loads, thermally driven air conditioning technologies such as desiccant dehumidification and cooling are promising options.

1.1. ECOS process description

As state of the art desiccant evaporative cooling systems of rotor type are mainly designed for higher air flow rates than needed for domestic use, a novel desiccant evaporative cooling cycle allowing a more effective use of the desiccant and designed for small-scale applications is subject to this paper. The main component of the novel desiccant evaporative cooling system is the **E**vaporatively **CO**oled **S**orptive-coated cross-flow heat exchanger (ECOS). In this air-to-air plate heat exchanger the primary side is coated with an adsorption material. Ambient air is dehumidified by adsorption of water vapor onto this desiccant. The heat exchanger walls of the primary channels are in thermal contact with the secondary cooling channels which are passed by building return air. As water is sprayed into the cooling channels, their heat exchanger walls are covered by a thin water film. This enables evaporation into the building return air passing the cooling channels. Therefore, adsorption heat is transferred to the cooling air side, leading to a temperature decrease of the sorption material. In case a higher amount of water can be evaporated than adsorbed, this will also lead to a temperature decrease of the supply air. The lower mean temperature of the sorption material shifts the sorption equilibrium towards higher loads. Thereby, this concept enables an enhanced dehumidification of ambient air coupled with its substantial sensible cooling [3].

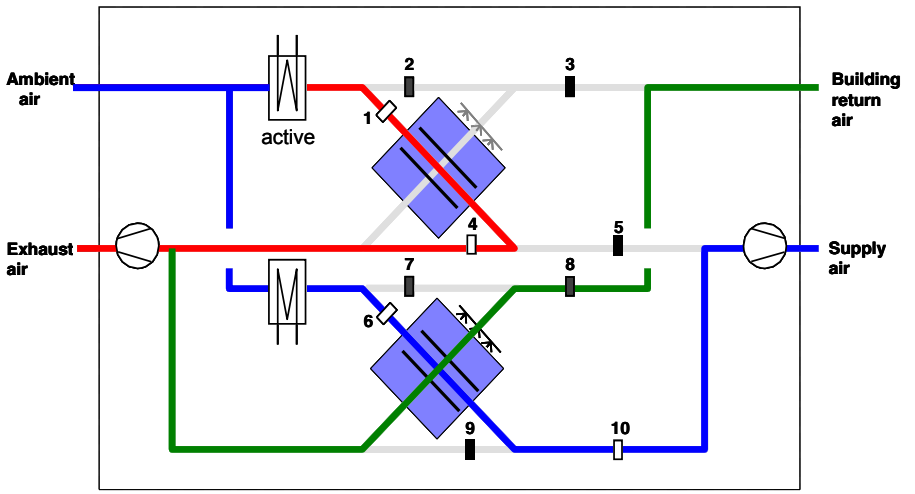

Fig. 1. Schematic of the ECOS system showing one heat exchanger in adsorption mode and the other heat exchanger being regenerated

Applying the concept in a ventilation device able to continuously supply air to a building, this leads to the batch-process configuration shown in Fig. 1. While the lower heat exchanger is in adsorption mode, producing the cooled and dehumidified supply air, the upper heat exchanger is regenerated by ambient air heated by the solar thermal source. The operation strategy of the two heat exchangers constituting the ventilation device is shown in Fig. 2 (top). While heat exchanger B is in adsorption mode, heat exchanger A undergoes first a desorption and then a short pre-cooling stage. During the pre-cooling stage, ambient air is flowing through the cooling channels of the heat exchanger. Due to simultaneous water spraying, evaporative cooling leads to a very fast and effective temperature reduction of the heat exchanger. The heat exchanger A then switches to adsorption mode, whereas heat exchanger B then switches to the desorption and subsequent pre-cooling stages. This way, a continuous supply of dehumidified and cooled air to the building is guaranteed.

Results presented in the following are derived from experiments and simulation of a single sorptive coated heat exchanger operating subsequently between adsorption, desorption and pre-cooling mode.

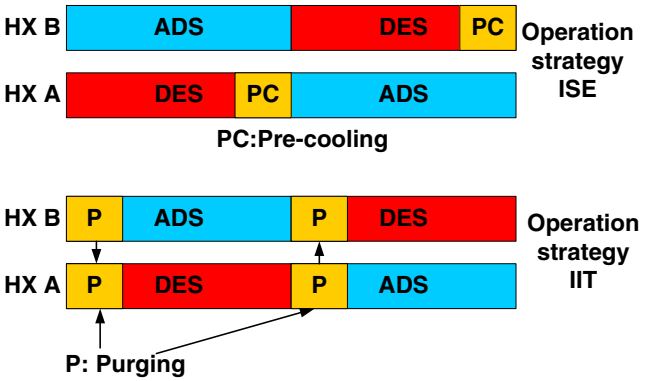

Fig. 2. Operation strategies of subsequent cycle stages for a ventilation device consisting of two sorptive coated heat exchangers. Upper schematic: currently followed approach, Lower schematic: approach formerly followed at IIT

1.2. Review of earlier literature

A similar air-cooled sorption cycle was first subject of research at the Illinois Institute of Technology (IIT) in the early 1980s [e.g. 4,5]. An air-to-air plate heat exchanger was covered with a desiccant material, which was then a teflon-silicagel-matrix glued to the heat exchanger base plates. However, the working principle was somewhat different from the one followed by our current research.

According to the above described ECOS principle, cooling of the heat exchanger, the desiccant and the process airflow occurs due to evaporation from the water-covered surface inside the cooling channels of the heat exchanger. In earlier research at IIT however, cooling of the heat exchanger was reached by heat exchange with a cold air stream. The latter may have been evaporatively cooled upstream of the cooling channels, therefore reaching a low temperature of the air at the inlet to the heat exchanger. The heat transfer mechanism inside the heat exchanger, however, was that of mere sensible cooling. This is also true for the more recent work of Weixing [6]. Therefore, outlet sorption side temperature always exceeded inlet temperature. Thus, no sensible cooling was achieved.

A second significant difference can be found in the operation strategy, displayed in Fig. 2 (bottom). According to the operation strategy followed at IIT, the supply of dehumidified air was not foreseen continuously. Instead, during an intermediate purging stage, heat was recovered from the just regenerated heat exchanger A to preheat the heat exchanger B before the latter would enter the regeneration stage. Thermodynamically, this is sensible as, firstly, regeneration heat is recovered and as, secondly, this allows more flexibly choosing the duration of the regeneration and adsorption stages. From an engineering point of view, however, this adds significantly to the complexity of the ventilation device, and from a user point of view, the risk of a penalty in comfort due to non-continuous air supply is given. The approach is therefore not followed in present research.

This paper aims at highlighting the benefit in performance arising from the internal evaporative, therefore latent, cooling in the heat exchanger. Further, a simulation based comparison between the earlier and the more recent prototype will be given for a measurement of air cooled adsorption.

2. Experimental analysis

An experimental facility was set up to evaluate the performance of the evaporatively cooled open sorption process. The experimental setup, prototypes tested and selected measurement results will be described in the following.

2.1. Experimental setup

The measurement setup designed to experimentally characterize a single sorptive coated heat exchanger operating according to the ECOS operation principle is schematically shown in Figure 3.a). The bold arrows indicate airflows through the sorptive coated heat exchanger during adsorption mode. Both cooling and sorption sides are passed by airflows. Upon switching to desorption mode, the flaps on the cooling side are closed and the flaps to the desorption side are opened. Only hot regeneration air flows through the sorption side during regeneration. During pre-cooling the flaps to the cooling side are opened again, whereas the flap to the heat exchanger on the sorption side is closed. Only the cooling side is passed by an airflow during the pre-cooling stage. In case of operation with evaporative cooling, water spraying is active during the adsorption and pre-cooling stages.

Measurement positions are also marked in Figure 2. Temperatures are measured with Pt100 sensors with an accuracy of ±0.3K. Humidity on the sorption side is determined by measuring dew point temperature using dew point sensors (IL Metronic) with an accuracy of ±0.5 K. Absolute pressure is measured with an accuracy of ±1mbar and differential pressure with an accuracy of ±2 Pa. Airflow rate on the sorption side is measured by an orifice plate with a relative accuracy of ±2% (range 100-550 kg/h). On the desorption and cooling side, airflow rates are subject to somewhat higher uncertainty with accuracies of ±4% (cooling side) and ±6% (desorption side).

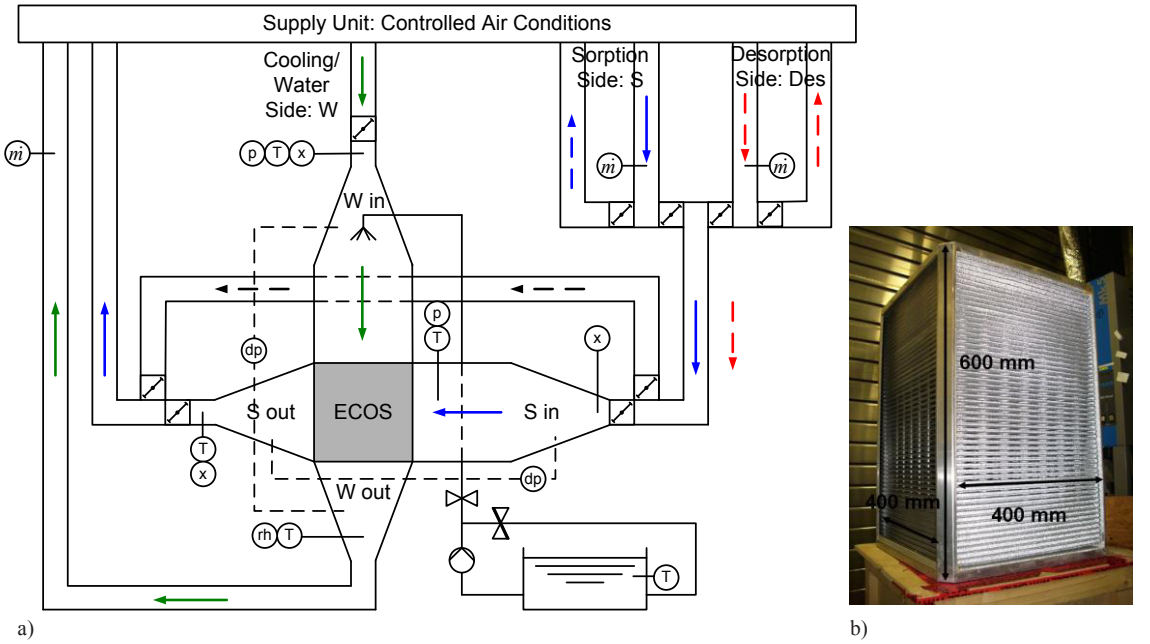

Fig 3. a) Schematic of the experimental setup; b) Sorptive coated heat exchanger prototype

2.2. Prototype dimensioning

The sorptive coated prototype subject to this paper is shown in Figure 3.b). Silicagel beads (Grace 123B) are fastened to the heat exchanger surface by an epoxy-coating-technique. The coating technique was developed by the company SorTech AG. The sorptive coated heat exchanger has an overall effective heat and mass transfer area of 16.42 m² and a total weight of 20.23 kg which includes a silica gel mass of 7.88 kg. As the humidity and temperature during the weighing of the silica gel mass were not recorded, the exact dry mass of silica gel is unknown. From uptake measurements to equilibrium and sorption equilibrium data, the total dry desiccant mass is estimated to approximately 7.4 kg.

Dimensions of the ISE prototype are summarized in Table 1. The latter also includes details of the earlier IIT-prototype which are taken from [4]. Data is given for means of comparison. The earlier prototype was of significantly bigger dimensions in terms of heat exchanger surface area, desiccant dry mass and heat exchanger total mass. As it was also operated with higher air flow rates, the mass flow rate normalized to surface area and dry desiccant mass is also given. From these figures it can be deducted, that the operation of the ISE prototype can be best compared to those of the earlier IIT prototype at higher air flow rate (1585 kg/h – 0.44 kg/s). Therefore, recorded data for the higher airflow rate tabulated in [4] will be applied for a prototype comparison via simulation.

Table 1. Dimensioning of sorptive coated heat exchanger prototypes

Parameter	Unit	Prototype IIT [4]	Prototype ISE
Outer Height/ Length/ Width	m	0.6/ 0.6/ 0.6	0.4/ 0.4/ 0.6
Number of channel pairs	-	81	54
Heat Exchanger Surface	m²	59	16.4
Sorption material	-	Silicagel-Teflon-Matrix	Silicagel Grace 123B (beads)
Thickness sorption material	mm	1.7	max 1.2
Free flow width sorption channel	mm	1.4	3.5
Mass Sorption material	kg	33.2: dry mass	7.9 (7.4: dry mass)
Total mass heat exchanger	kg	74.5	20.2
Desiccant mass fraction	-	0.45	0.39
Nominal sorption air mass flow rate	kg/h	648 / 850 / 1584	450
Mass flow rate/ Surface area	kg/(h m²)	11.0 / 14.4 / **26.8**	**24.4**
Mass flow rate/ Desiccant mass	kg/(h kg)	19.5 / 25.6 / **47.7**	**50.6**
Flow velocity sorption channel	m/s	2.2 / 2.9 / 5.4	1.45
Re-Number sorption channel	-	380 / 500 / 910	630

2.3. The effect of evaporative cooling in the sorptive coated heat exchanger

In order to assess the effect of evaporative cooling in the sorptive cooled heat exchanger, experiments with nearly equivalent inlet conditions were conducted. Figure 4 shows the measured temperature (a) and humidity ratio (b) time series on the sorption side of six measured sorption cycles – three cycles subject to air cooling (AC) and further three cycles subject to evaporative cooling (EC). Due to the good repeatability of the measurements, the temperature and humidity ratio evolutions are almost identical for the cycles which are cooled by the same method (AC or EC). Inlet conditions of the experiment are given in Table 2.

The time evolutions show the three subsequent adsorption stages pre-cooling, adsorption and desorption. During pre-cooling there is no air flow through the sorption side. Temperatures and humidity ratios measured in the still air volume are therefore only indicative. During pre-cooling the temperature falls as is true for the humidity ratio. As the desiccant is cooled, water vapor from the air volume is adsorbed, leading to this decrease in humidity ratio. Adsorption is then strongest and therefore outlet humidity ratio lowest directly after switching to the adsorption stage. In the long cycle shown here, saturation is almost reached at the end of the adsorption stage, as outlet humidity converges towards inlet humidity ratio. It is obvious that outlet humidity ratio during the adsorption stage is always lower for the evaporatively cooled cycle as opposed to the air cooled cycle, leading to a higher amount of water adsorbed. During desorption, outlet humidity ratio rises over inlet humidity ratio. This is more pronounced for the evaporatively cooled cycle in which the higher amount of water adsorbed must also be desorbed.

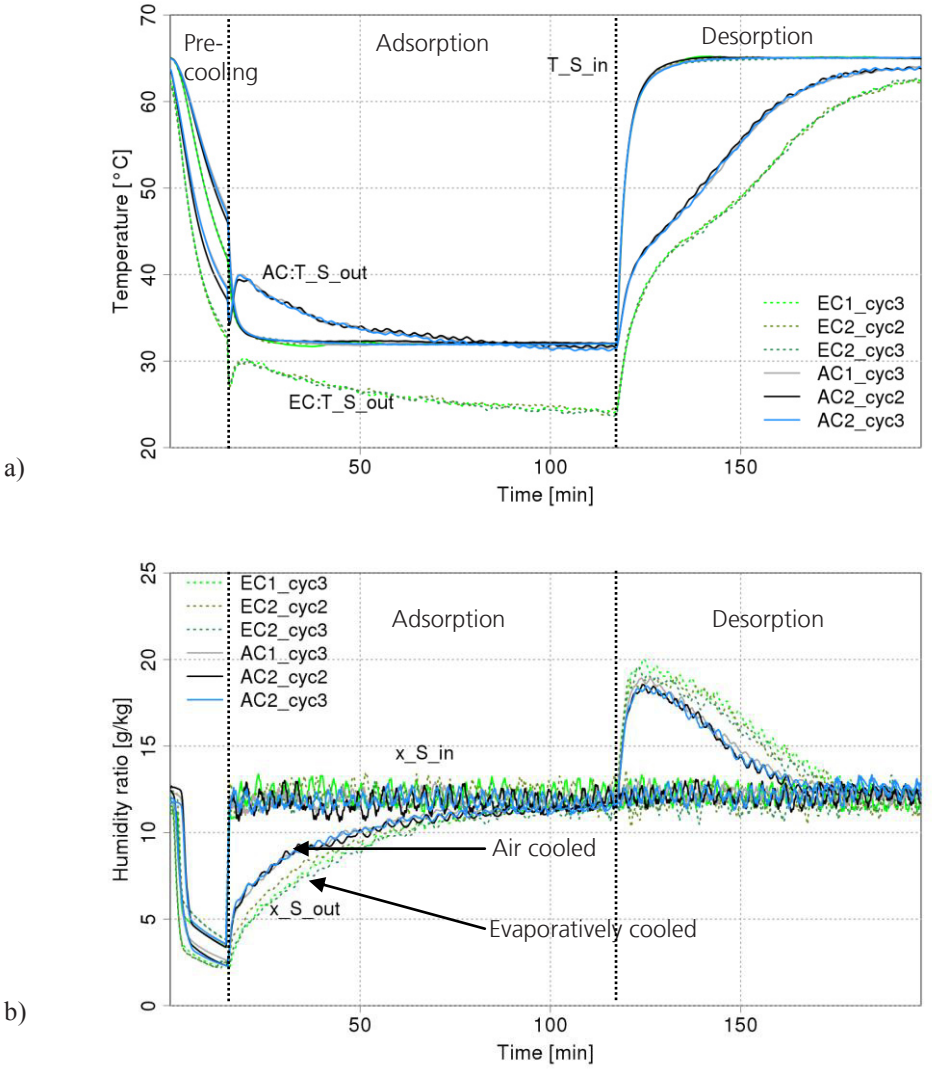

Fig. 4. a) Measured temperature and b) measured humidity ratio time series of long sorption cycles with air cooling and evaporative cooling

Outlet temperature corresponds to the outlet humidity ratio time series. During the beginning of adsorption, the outlet temperature is highest due to the strongest release of adsorption heat. Due to evaporative cooling, a quasi parallel shift to lower outlet temperatures by approximately 5 K can be observed. It is remarkable, that the outlet temperature is always below the inlet temperature, indicating that the entire adsorption heat and additional sensible heat can be removed by evaporative cooling in the heat exchanger.

To further evaluate performance the following performance figures are introduced:

- **Adsorbed water mass** [kg_W]: Total water uptake during the adsorption stage
- **Cooling capacity** [W]: Cooling power calculated from the mean enthalpy difference between inlet and outlet of the sorption side during the adsorption stage
- **Moisture cycled** (from [4]) [kg_W/kg_D]: Adsorbed water mass normalized to the desiccant dry mass
- **Mean dehumidification** [kg_W/kg_A]: Mean difference between inlet and outlet humidity ratio during adsorption

Table 2. Inlet conditions of the two measurements of long adsorption cycles

Inlet conditions	Unit	Air cooled long cycle	Evaporatively cooled long cycle
T inlet, sorption side	°C	32.3-32.4	32.2-32.3
x inlet, sorption side	g/kg	11.8	11.8-12.2
T inlet, cooling side	°C	30.0-30.4	30.1-30.2
x inlet, cooling side	g/kg	12.1	11.7-12.2
All mass flow rates	kg/h	450	450
Regeneration inlet temperature	°C	65.0-65.1	65.0

The performance indicators for the shown experiments are given in Table 3. In comparison, the evaporatively cooled sorption cycle outperforms the air cooled sorption cycle, with an increase in adsorbed water mass by 46 % and an enhancement of the cooling capacity by factor 4.1. Moisture cycled also increases, showing the more effective use of the desiccant in the evaporatively cooled sorption cycle.

Table 3. Performance figures of the two measurements of long adsorption cycles

Performance	Unit	Air cooled long cycle	Evaporatively cooled long cycle
Adsorbed water mass	kg_{Water}	1.3 ± 0.2	1.9 ± 0.2
Cooling Power	W	380 ± 120	1560 ± 120
Mean dehumidification	kg_{Water}/kg_{Air}	1.7 ± 0.3	2.5 ± 0.3
Moisture cycled	kg_W/kg_S	0.16 ± 0.05	0.24 ± 0.06

3. Simulation based comparison of prototypes

As inlet conditions in the measurements of the air cooled IIT prototype and the air/evaporatively cooled ISE prototype are very different, a validated dynamic simulation model [7] is applied with inlet data taken from [4, Table 7.1] for prototype comparison. The model, implemented using Modelica/ Dymola, was also validated with high accuracy for the above described measurements. For the air cooled cycles, deviations between measurement and simulation are < 2 % for adsorbed water mass and in the range of 4 - 7 % for cooling capacity.

To compare prototypes, input data was generated from the tabulated measurement of air cooled adsorption in [4]. As data is not given for the purging phase, conditions and length (5 minutes) of the

purging phase had to be estimated from the information given. Simulations were then performed for the air cooled process with the model parameterized according to the ISE prototype. To ensure the equivalent ratio of mass flow rate to desiccant mass (47.7 kg/ ($s \cdot kg_S$)in IIT data), the inlet mass flow rate to the simulation of the ISE prototype was set to 350 kg/h.

Measured and simulated humidity ratio time series for the air cooled adsorption are shown in Figure 5. The inlet and outlet humidity ratios, derived from measurement data of Worek [4], are shown as white and black triangles. The measured outlet humidity ratio is lowest at the start of adsorption and then rises, similarly as described before. Mean measured dehumidification is $\Delta x = 3.8$ g/kg. In the baseline simulation, however, a different curve shape can be observed. Outlet humidity ratio remains at a fairly constant level. Overall, a higher mean dehumidification of $\Delta x = 4.9$ g/kg than measured is calculated from the baseline simulation of air cooled adsorption. In simulation and measurement the outlet temperature is higher than the inlet temperature (not shown). Assuming evaporatively cooled adsorption (not shown; inlet conditions cooling side: 23.3 °C/13 g/kg) mean dehumidification can be increased to $\Delta x = 5.4$ g/kg and a sensible cooling effect is achieved, thus outperforming the air cooled cycle.

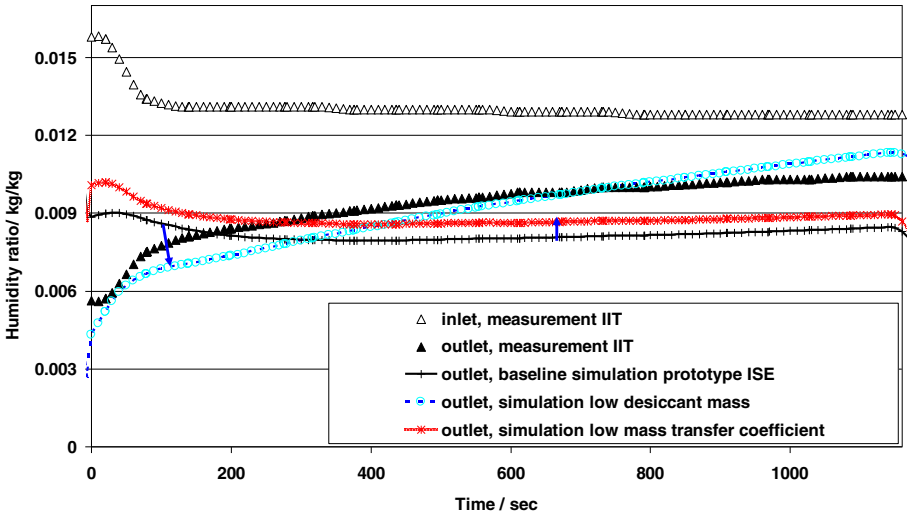

Fig. 5. Humidity ratio time series measured at IIT [4], calculated from simulation with base configuration, reduced mass transfer coefficient and reduced desiccant mass: the curve shape indicates only partial active use of the desiccant in the former IIT prototype.

As equilibrium data used for calculations matches well the data given in [4], the mean dehumidification measured, which is lower than expected from the simulation, may be explained by a different effectiveness of mass transfer processes in both prototypes. The flow velocity in the earlier IIT prototype is about four times higher (see Table 1) than in the ISE prototype, leading to much shorter contact times between the desiccant and the airflow. Further, it is possible that only a fraction of the desiccant mass is actively cycled, which is also indicated by the lower figure of moisture cycled of 0.06 for the given measurement (around 0.11 in ISE prototype measurements of shorter cycles).

In the used pseudo-gas side resistance model, the effectiveness of the mass transfer process is reflected by an effective mass transfer coefficient β_S which represents both gas and solid side resistances to mass

transfer. Lowering this effective mass transfer coefficient β_S by 50% in the simulation leads to the almost parallel shift of the outlet humidity ratio time series to higher humidity ratios, but does not alter its shape. Lowering the desiccant dry mass by factor 0.25 in the simulation instead leads to the steeper curve shape, similar to the one measured. This again indicates that in the earlier IIT prototype, not the entire desiccant dry mass was actively cycled, highlighting that maximizing the desiccant mass alone does not necessarily lead to a higher dehumidification performance. As such behavior was not observed for the ISE prototype, it can be deducted that a higher fraction of the sorption material is active in ad/desorption. A reason for this could be the higher effective mass transfer area due to the coating with silica gel beads. The circular bead surface makes the desiccant more accessible to water vapor and reduces the required length of vapor diffusion in the desiccant. Analyzing this in more detail could be subject of future research, e.g. a detailed diffusion model could be implemented for different surface structures and desiccant coatings.

4. Conclusions

The evaporatively cooled adsorption cycle revealed a clear advantage in performance compared to the air cooled process in both measurements and simulations. For the presented long adsorption cycle adsorbed water mass could be increased by 46% and cooling power by a factor of 4.1. Evaluating earlier research with a model validated with recent data showed that the amount of desiccant active in adsorption is crucial to achieve high dehumidification rates. The higher fraction of desiccant cycled in the newer ISE prototype and the internal evaporative cooling lead to a promising advance in overall performance.

Acknowledgements

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References

[1] Gignon-Masse L, Riviere P, Adnot J. Strategies for reducing the envrionmental impacts of room air conditioners in Europe. *Energy Policy* 2011;**39**: 2152-2164.

[2] Riviere P, Adnot J, Grignon-Masse L, Legendre S, et al. Preparatory study on the environmental performance of residential room conditioning appliances (airco and ventilation) – Economic and Market Analysis, ECODESIGN Lot 10 Draft of Chapter 2, Draft report of Task 2. Study for the D.G. Transportation –Energy (DGTREN) of the Commission of the E.U. 2008.

[3] Motta M, Henning H-M. A novel high efficient sorption system for air dehumidifications (ECOS). *Proceedings International Sorption Heat Pump Conference* (ISHPC). Denver, Colorado, June 2005.

[4] Worek WM. Experimental performance of a cross-cooled desiccant dehumidifier prototype, PhD thesis, Illinois Institute of Technology, Chicago, IL, 1980.

[5] Monnier JB. Analysis of desiccant cooling systems using advanced desiccant matrix structure, PhD thesis, Illinois Institute of Technology, Chicago, IL, 1986.

[6] Weixing Y, Yi Z, Xiaoru L, Xiugan, Y: Study of a new modified cross-cooled compact solid desiccant dehumidifier. *Applied Thermal Engineering*; **28**:2257-2266.

[7] Bongs C, Morgenstern A, Lukito, Y, Henning H-M. Performance Analysis and Model Validation of an Evaporatively Cooled Sorptive Heat Exchanger (ECOS). *Proceedings OTTI 4th International Conference Solar Air-Conditioning*, Larnaka, Cyprus, Octobre 2011.